PATENT SPECIFICATION

944,723

DRAWINGS ATTACHED.

944,723

Date of Application and filing Complete Specification: *Nov. 24, 1960.* *No. 40454/60.*

Application made in United States of America (No. 862,404) on Dec. 28, 1959.

Complete Specification Published: Dec. 18, 1963.

© *Crown Copyright 1963.*

Index at Acceptance:—**Class F4** S2B.

International Classification:—F 25 h.

COMPLETE SPECIFICATION.

Improvements in or relating to Tubes for Steam Condensers.

We, CALUMET & HECLA, INC., a corporation organized under the laws of the State of Michigan, United States of America, having a place of business at 17200 Southfield Road, Allen Park, Michigan, United States of America, do hereby declare the invention, for which we pray that a patent may be granted to us, and the method by which it is to be performed, to be particularly described in and by the following statement:—

The present invention relates to tubes for steam condensers.

In certain applications a prime consideration governing the construction of a steam condenser is the space occupied by the condenser. To provide the most efficient condenser in the smallest possible space it is essential to provide condensing tubes having an overall heat transfer coefficient as high as possible. From purely geometrical and practical considerations it is found that the height and thickness of fins provided on the exterior surface of the water tubes should be limited, not to exceed $\frac{1}{4}$ of an inch in height above the outside or fin root diameter of the tube, and not to exceed $\frac{1}{4}$ of an inch in mean thickness.

In general, the provisions of fins at the outer surface of the tube increases the area for heat transfer between the steam and the water tube or its contents. However, the selection of the fin height, fin thickness, and fin spacing for producing the most efficient heat transfer requires consideration of a number of factors other than a simple increase in external surface area of the tube.

In considering overall efficiency of the tube in effecting transfer of heat from the steam to the condensing water within the tube, it is of course essential to consider the heat transfer characteristics at the inside surface of the tube. The transfer of heat through the inner surface of the tube can be increased in the first place by increasing the area per unit length of the inner surface of the tube, and in the second place by increasing the turbulence level inside the tube with no increase in volumetric quantity flow.

From practical considerations the deformation of the inner surface of the tube will be in large part controlled by the fin structure provided on the exterior surface of the tube. Thus, fins may be provided which are annular and a multiplicity of separate fins may be provided. Alternatively, one or more helical fins may be provided with the adjacent convolutions thereof spaced in accordance with requirements which will be discussed hereinafter. In any case, where an outstanding extending fin is provided on the exterior surface of the tube there will be an opposed corresponding groove or channel at the interior surface of the tube. The groove or channel will of course have a pitch corresponding to that of the fin. However, the groove or channel and the resultant raised ridges intermediate the grooves or channels may vary somewhat as to mean thickness of the ridge, mean width of the groove or channel (or the spacing between adjacent ridges or convolutions thereof), and the mean height of the ridges.

Although in an actual tube, the fins may have somewhat different cross-sectional shapes, the heat transfer effectiveness of the tube may be determined by calculations based upon the assumption that the external fins are of rectangular cross-section. It is also assumed herein that the height of the ridges at the inner surface of the tubes, as measured between the crest of a ridge and the bottom of the adjacent grooves or

channels is substantial, for example not less than 25% of the height of the external fins. In any case of course, it should not exceed the height of the external fins. For convenience in calculation the internal ridges are assumed to be of semi-circular cross-section, but results will not be substantially different if the internal ridges have different shapes as specifically disclosed herein.

It has been found that the optimum tube shape depends among other things on the pressure in the condenser. At high vacuum conditions, in general a larger fin height to fin spacing ratio (S/H) is desirable.

According to the present invention there is provided a single wall metal tube for use in a steam condenser adapted to have cooling water circulated therethrough to condense steam on its exterior surface, said tube having at its inner surface a multiplicity of uniformly axially spaced generally circumferentially extending ridge portions separated by grooves or channel portions, the ridge portions having an axial spacing at least three, and preferably as much as five, times as great as the radial height thereof, the internal ridge portions having a height of .010—.030 inches and an axial spacing of .05—.16 inches.

For a better understanding of the present invention and to show how the same may be carried into effect, reference will now be made, by way of example, to the accompanying drawings, in which :—

Figure 1 is a fragmentary axial section of a finned water tube.

Figures 2 and 3 are views similar to Figure 1, illustrating variations in the shape of the inner tube surface.

Figure 4 is a fragmentary section of a tube to which are applied various dimensional identifying letters used in calculations herein.

Figures 5A, 5B, 5C and 5D are graphs showing the effect of variation in number of fins per inch in the effective external tube area.

Figure 6 is a graph showing the effect of spacing of internal ridges on the heat transfer coefficient of the internal tube surface.

Figures 7A, 7B, 7C, 7D and 7E are graphs showing variations in the overall resistance factor of the tube as influenced by the space-height ratio of the external fins.

Figures 8A and 8B are graphs showing the proper space-height ratio for different tube designs for most efficient overall operation.

Figures 9—14 are enlarged fragmentary sectional views showing a variety of specifically different tube designs.

In commercial steam condensers, cold water is passed through tubes and steam is condensed on the outside surface of the tubes. The performance of steam condensers, as measured by the heat transfer rate per unit length of tubing, depends on a number of factors including the tube configuration.

This discussion concerns tube configuration as it effects the tube surface area and the turbulence of the coolant fluid. Heat transfer performance usually is improved by increasing the tube surface area and by increasing turbulence in the fluids involved. However, it will be shown that the processes which are usually thought to improve tube performance can actually be deleterious under some circumstances and that there is an optimum in the general sense.

Basic Heat Transfer Relationships.

The basic relationship governing heat transfer in condensers may be written

$$\frac{Q}{TD} = \frac{1}{\dfrac{1}{h_o A_o} + \dfrac{1}{h_i A_i} + r} \quad (1)$$

where $\frac{Q}{TD}$ is the heat transfer rate per mean temperature difference for a tube of unit length, Q being the quantity of heat transfer and TD being the temperature difference between the two fluids ;

h_o is the heat transfer coefficient at the tube outer surface ;

h_i is the heat transfer coefficient at the tube inner surface ;

A_o is the tube external surface area per unit length ;

A_i is the tube internal surface area per unit length ; and

r is a factor which includes the resistance to heat transfer of the tube metal and of any dirt or scale which may be deposited on the tube.

The factor r is fixed by the condenser service conditions including operating pressure, operating temperature, and cleanliness of the fluids involved. As a result, this factor may not be varied at will, and it may be considered a constant independent of tube configuration.

The factors $1/h_oA_o$ and $1/h_iA_i$ do depend on tube configuration. These factors may be thought of as resistances to heat transfer which should be minimized.

External Tube Performance.

In steam condensing, liquid condensate is formed on the outer surface of the heat transfer tube. This liquid condensate may be retained on the tube surface by surface tension effects where it acts to a large degree as an insulator.

If the fins on an extended surface tubing are relatively close together the surface may be completely covered with liquid except at the maximum fin periphery. This condensate retention is referred to in the heat transfer industry as "flooding."

The water tubes in the steam condenser will of course be positioned horizontally or substantially horizontally so that condensate formed on the exterior surfaces of the tubes and particularly between the fins thereof, may most readily drain off the tubes. Under certain circumstances condensate may flow to the lower portion of the space between adjacent fins or convolutions of fins and be retained therein in a generally arcuate zone subtending an angle having its apex at the centre of the tube.

The fraction of the surface which is covered by liquid can be calculated by a simple force balance. As an illustration, consider the case of condensate retention between the fins with rectangular cross-section as shown at the bottom of Figure 4. By a force balance, it can easily be shown that

$$\frac{a}{\sin a} = \frac{8t}{d\ g}\left[\frac{2D_o - D_r + S}{(D_o^2 - D_r^2)\ S}\right] \qquad (2)$$

where a is one-half the angle subtended by the arcuate zone of liquid between the fins;
d is the density of the condensate;
t is the surface tension of the condensate;
g is the acceleration due to gravity;
D_o is the fin outer diameter;
D_r is the fin root diameter; and
S is the space between fins in the direction of the tube axis.

Suppose a tube is of constant dimensions except for S, the space between fins. For large S, no liquid is retained between the fins. As S is reduced, a will increase until $a=\pi$ radians (or $a=180°$) at which point liquid completely fills the space between the fins. At any spacing a/π is the fraction of the surface area (excluding fin tip area) which is occluded by condensate.

In computing the heat transfer resistance, the liquid retention effect must be taken into account. As was mentioned earlier, the external resistance is given by $1/h_oA_o$. To account for liquid retention, A_o will be replaced by A_e which is defined to be the effective surface area, the surface area not occluded by condensate.

Effective surface areas have been caculated for tubes of several different dimensions as shown in Figure 5. Conclusions which may be drawn from these calculations include:—

1. The effective heat transfer surface area increases linearly with increasing number of fins per inch only up to the point of incipient liquid retention. The effective surface area decreases rapidly beyond this point because of liquid retention despite the fact that the total surface area is increasing.

2. An effective area near the maximum is obtained for only a very narrow region of fin spacing in all cases of $\frac{1}{4}$ inch fin height. For lower fin heights, the maximum becomes more nearly flat so that fin spacing becomes a less crucial variable.

Internal Tube Performance.

It is well known that turbulence in a fluid stream flowing in a tube promotes heat transfer. It is also known that transverse curvature of the tube wall (or transverse internal ridges) will cause higher turbulence than would be expected from a smooth, cylindrical tube. Transverse curvature of an inner tube wall has the additional important effect of increasing the surface area per unit length for heat transfer over that of a smooth, cylindrical tube. However, as will be discussed below, the heat transfer performance of a tube is not increased indefinitely by indefinitely increasing the number of transverse ridges. There is an optimum beyond which increasing the number of transverse ridges is deleterious.

As an illustration, consider internal transverse ridges of semi-circular cross-section as indicated at 18 in Figure 4. If the ridges are a great distance apart the tube will tend to perform as a plain tube. As the ridge spacing is reduced, turbulence in the flowing stream is increased which in turn increases the heat transfer coefficient. However, when the ridge spacing is sufficiently reduced, circulation of the fluid between the ridges is hindered and the ridges become less effective. The fluid between the ridges becomes essentially stagnant. In the limit, if the ridges are such that no space at all exists between them, the inner tube surface is once again a plain surface. In short, there is a point of maximum performance with respect to ridge spacing.

The effect of ridge spacing on performance has been investigated experimentally. Figure 6 shows the ratio $A_ih_i/(A_ih_i)_p$, where the subscript p denotes a tube having a plain interior surface, as a function of the ridge spacing to height ratio. The conclusions from this work include:—

1. The performance ratio, $A_ih_i/(A_ih_i)_p$, is as much as 3.6 at the maximum point. That is to say, an internally ridged tube may have less internal resistance to heat transfer than a plain tube by the factor .36.

2. The Reynolds' number of the fluid does not have much effect on the performance ratio, at least in the range of practical importance. This statement is equivalent to saying the performance ratio is approximately independent of fluid bulk velocity or viscosity. It is understood that the plain and internally ridged tubes are compared at the same fluid bulk velocity and viscosity.

Optimum Tubing.

Consider tubing which has circumferential

external fins and internal ridges. The fins and ridges discussed here are understood to be at least partly transverse. That is to say that strictly longitudinal fins and ridges are excluded, whereas helical and strictly transverse fins and ridges are included.

For convenience, the resistance due to the external coefficient h_o and internal coefficient h_i, as indicated in equation (1) will be combined to give the heat transfer resistance R which is dependent on tube configuration :—

$$R = \frac{1}{A_e h_o} + \frac{1}{A_i h_i} \quad (3)$$

Equation (3) may be multiplied by $(A_i h_i)_p$ to give

$$R^1 = R(A_i h_i)_p = \frac{(A_i h_i)_p}{A_e h_o} + \frac{(A_i h_i)_p}{A_i h_i} \quad (4)$$

The optimum tube is clearly one for which R^1 is a minimum. The quantities $(A_i h_i)_p / A_i h_i$ and $A_e/A_i)_p$ can be evaluated for any given tube by the methods discussed above. The factor $(h_i)_p/h_o$ may be taken to be a constant independent of tube configuration. This ratio $(h_i)_p/h_o$ is usually of the order of 2/3 for steam condensing. However, it also depends on the many condenser design and operating variables other than tube configuration.

Values of R^1 have been computed for a number of tubes as shown in Figures 7A to 7E. The values were computed for $h_i)_p/h_o = 1$, and for $(h_i)_p/h_o = 0.5$ to cover the practical range. Note that in each case there is a minimum R^1 at some value of fin spacing to fin height ratio (S/H). The optimum cannot be expected to be attained exactly in commercial practice. For this reason, an "optimum range" will be considered. The optimum range of S/H ratios is defined to be the region of S/H values for a given tube configuration where R^1 is within ten per cent of the minimum R^1.

In Figures 7A to 7E it will be observed that the several different curves are drawn for different values of fin thickness of T and H. In each of these curves there is included a cross-section of a fin having the assumed dimensions of height and thickness.

The optimum range of S/H ratios is presented in Figures 8A and 8B as a function of tube configuration variables. A single lower line represents the minimum limit of S/H values in the optimum range. The upper limit of optimum S/H is represented by a family of lines since the upper limit varies with tube diameter and fin thickness as well as with fin height.

The following conclusions may be stated:

1. For steam condensing, tubing with both internal and external extended surface is more effective than plain, smooth tubing.

2. There is an optimum range with respect to fin spacing and ridge spacing where tube performance is near the maximum.

3. The fin and ridge spacing to height ratio must always exceed 0.35 for optimum performance.

4. The fin and ridge spacing to height ratio must be less than some upper limit for optimum performance. This upper limit is given graphically in Figures 8A and 8B.

Referring now to Figure 4 there is shown a tube 10 having an outside or fin root diameter D_r, and an outside diameter D_o measured to the outside of the fins 12. The outside fins 12 have a mean width or thickness dimension T and a mean height H. The space 14 between adjacent fins or convolutions of fins has the dimension S. In this Figure the grooves or channels 16 at the inner surface of the tube 10 are separated by the ridges 18 which as will be apparent from the Figure are opposite the spaces 14 between adjacent fins 12.

As shown in Figure 1, the tube herein designated 20 is provided with fins 22 whose thickness is substantially equal to the width of the space 24 between adjacent fins. At the inner surface of the tube the grooves or channels 26 are defined by the inter-section between convexly curved ridges 28.

In Figure 2, while the external fins 22 and spaces 24 are as shown in Figure 1, the inner grooves or channels 36 are associated with the intermediate ridges 38 in such a way that a smooth continuous undulating interior surface is obtained.

In Figure 3, while again the fins 22 and spaces 24 are as shown in Figure 1, the grooves or channels 46 formed between the adjacent ridges 48 are relatively wider and have flat bottom portions 49 blending smoothly into the side surfaces of the ridges 48. In addition, the crests of the ridges 48 are shown as flat, a condition which results from carrying out the exterior finning operation with a cylindrical mandrel within the tube. By this operation accurate control of the final inside tube diameter is maintained.

The tube structure indicated in Figures 1, 2 and 3 are more or less diagrammatic and illustrate assumed or theoretical shapes.

Referring now to Figures 9—14 there are shown some actual cross-sections of condenser tubes which have been given practical tests in water tube steam condensers.

In Figures 11 and 12 the various dimensions of the tube have been designated by the symbols which are identified below :—

D_o Outside diameter measured at fin crests.

D_r Outside root diameter measured at the bottoms of the spaces between adjacent fins.

$D_{i\,max}$ Maximum internal diameter measured at the bottoms of the grooves or channels between adjacent internal ridges.

$D_{i\,min}$ Minimum internal diameter measured at the crests of the internal ridges.

S_o Spacing between adjacent external fins.

T_o Thickness of external fins.

H_o Height of external fins.

H_i Height of internal ridges.

fpi Fins per inch.

S_i Space between internal ridges.

T_i Thickness of internal ridges.

S_o/H_o Ratio of fins spacing to fin height.

The fins of Figures 9—14 have dimensions as set forth in the following tabulation :—

	Figure 9	Figure 10	Figure 11	Figure 12	Figure 13	Figure 14
D_o	0.617	0.617	0.613	0.7180	0.7226	0.7181
D_r	0.566	0.566	0.473	0.646	0.632	0.644
$D_{i\,max}$ ..	0.502	0.502	0.437	0.5427	0.5707	0.5453
$D_{i\,min}$ ..	0.471	0.471	0.376	0.5141	0.5191	0.5168
S_o	0.056	0.056	0.078	0.0831	0.0802	0.0837
T_o	0.161	0.070	0.075	0.0659	0.1154	0.0638
H_o	0.0252	0.0252	0.070	0.036	0.045	0.037
H_i	0.015	0.015	0.030	0.0143	0.0258	0.0142
fpi	4.85	4.85	6.65	6.7	5.1	6.8
S_i	0.154	0.085	0.095	0.0608	0.088	0.0569
T_i	0.042	0.042	0.058	0.0881	0.1076	0.0906
S_o/H_o ..	2.24	2.24	1.16	2.32	1.78	2.26

It will be noted that the ratio S_o/H_o varies from approximately 2.3 for the tube shown in Figure 12, to 1.2 for the tube shown in Figure 11. The tubes having the higher S_o/H_o ratios are particularly efficient under relatively high vacuum conditions. The tube of Figure 9, for example, was operated at 1.0 p.s.i.a. and gave substantially improved results as compared to the tube shown in Figure 12. On the other hand, under elevated pressure conditions such for example as 5.0 p.s.i.g., the tube of Figure 10 proved more efficient than the tube shown in Figure 9.

In all of the practical tubes shown in Figures 9—14 it will be observed that the internal groove or channel and ridge formation is such as will be produced by the formation of the external fins by tubular fin rolling. As a consequence, the average number of external fins per inch is of course the same as the average number of internal ridges per inch.

In the tabulated values for the several tube characteristics set forth above, it will of course be appreciated that there may be some variation in determining some of the dimensions. For example, in Figure 9 the dimension S_o may be measured from points spaced outwardly from the upper corners of the fins, as is also the dimension T_o. The average or mean fin spacing may therefore vary somewhat from the indicated value, and the values given are thus not precise.

An analysis of the shapes of the several different practical fin constructions indicates that for high vacuum condensers the S_o/H_o ratio should be greater than 1.0 and preferably between 1.0 and 3.0. In high pressure condensers, however, this ratio may safely approach the previously described lower limit of 0.35.

The tubes having the wall configuration illustrated in Figures 9, 10, 13 and 14 appear to be the most efficient when employed as water tube steam condensers. From the table above it will be observed that the approximate range of dimensions for these tubes is as follows :—

Mean space S_o between adjacent external fins	.05—.09"
External fin height H_o ..	.025—.045"
Internal ridge thickness T_o ..	.04—.110"
Mean channel or groove width S_i between internal ridges ..	.05—.16"
Internal ridge height H_i ..	.010—.030"

The drawings and the foregoing Specification constitute a description of the improved finned condenser tube in such full, clear, concise and exact terms as to enable any person skilled in the art to practice the invention, the scope of which is indicated by the appended claims.

WHAT WE CLAIM IS :—

1. A single wall metal tube for use in a steam condenser adapted to have cooling water circulated therethrough to condense steam on its exterior surface, said tube having at its inner surface a multiplicity of uniformly axially spaced generally circumferentially extending ridge portions separated by grooves or channel portions, the ridge portions having an axial spacing at least

three, and preferably as much as five, times as great as the radial height thereof, the internal ridge portions having a height of .010—.030 inches and an axial spacing of .05—.16 inches.

2. A tube as claimed in Claim 1, wherein the exterior surface of the tube has outwardly projecting fin portions in substantial alignment with the internal groove or channel portions therein.

3. A tube as claimed in Claim 2, wherein the external fin portions have a height of .025—.045 inches and an axial spacing of .05—.09 inches.

4. A tube for use in a steam condenser, the tube being substantially as hereinbefore described with reference to Figures 9, 10, 13 or 14 of the accompanying drawings.

5. A water tube steam condenser provided with a multiplicity of parallel tubes constructed as defined in any preceding claim.

HASELTINE, LAKE & CO.,
Chartered Patent Agents,
28 Southampton Buildings,
Chancery Lane, London, W.C.2.
Agents for the Applicants.

Abingdon: Printed for Her Majesty's Stationery Office, by Burgess & Son (Abingdon), Ltd.—1963.
Published at The Patent Office, 25, Southampton Buildings, London, W.C.2,
from which copies may be obtained.

FIG.1.

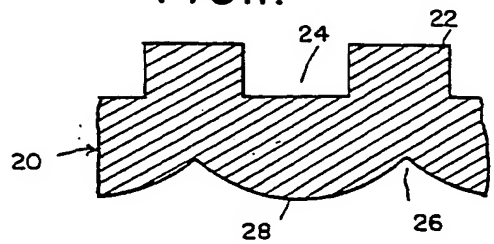


FIG.2

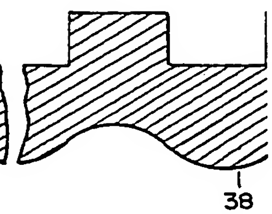


FIG.7A.

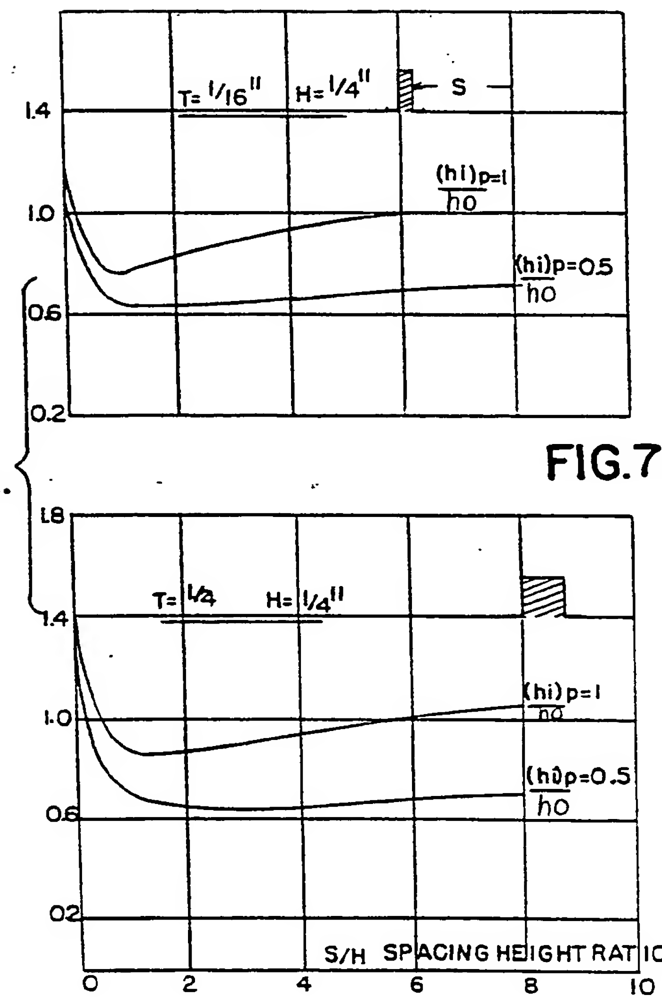
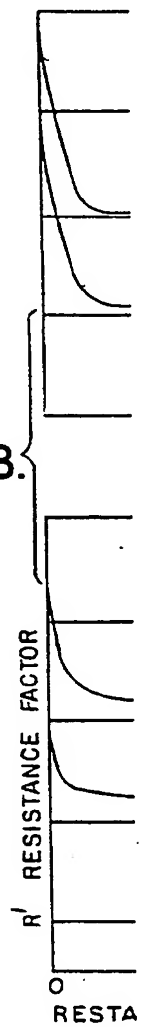


FIG.7B.



944723 COMPLETE SPECIFICATION

5 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 1

FIG.2.

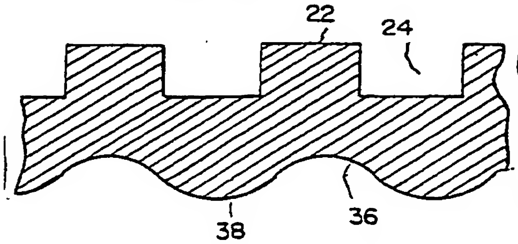


FIG.3.

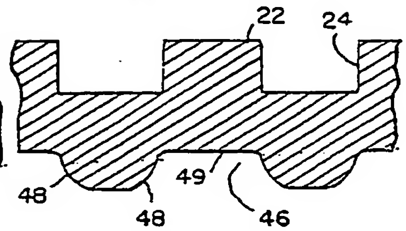
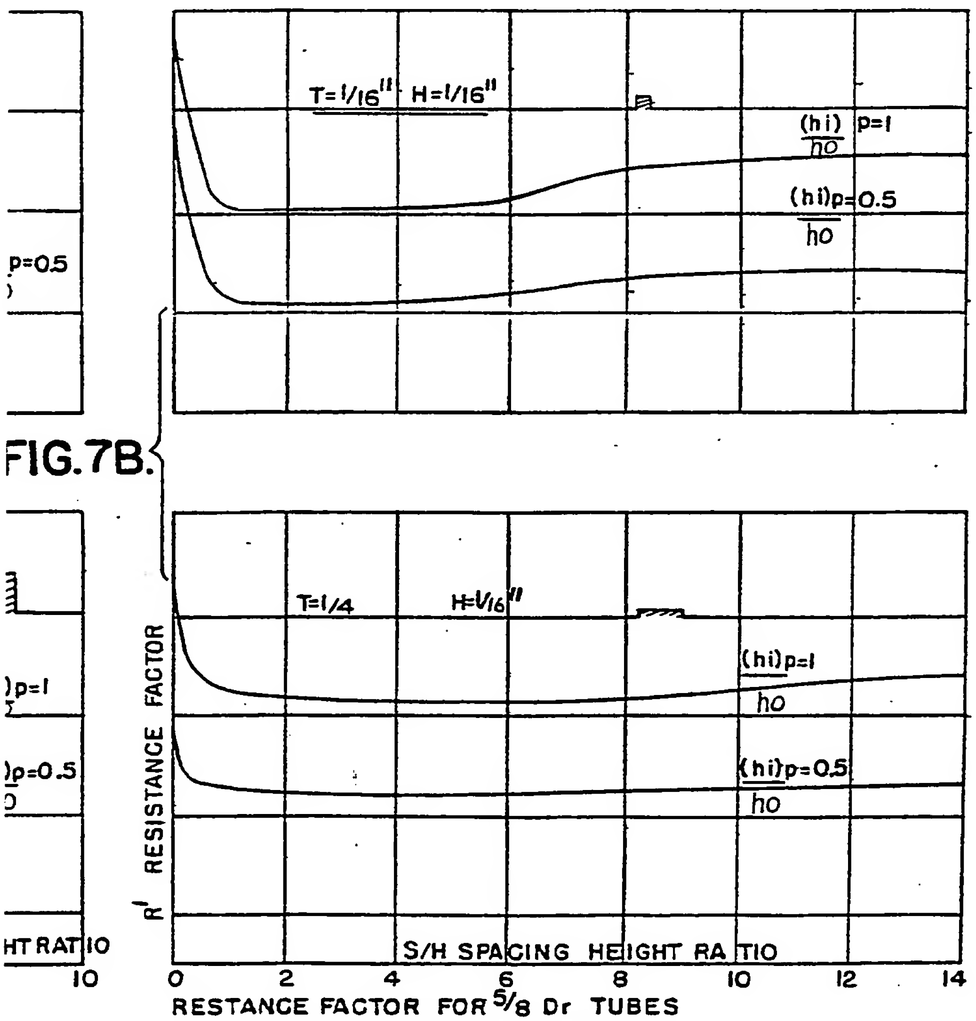


FIG.7B.



944723 COMPLETE SPECIFICATION
5 SHEETS This drawing is a reproduction of the Original on a reduced scale
Sheet 1

FIG.1.

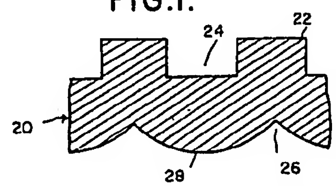


FIG.2.

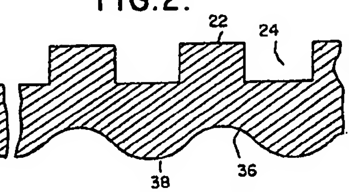
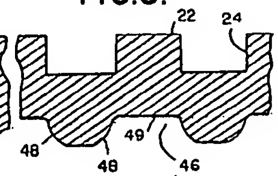


FIG.3.



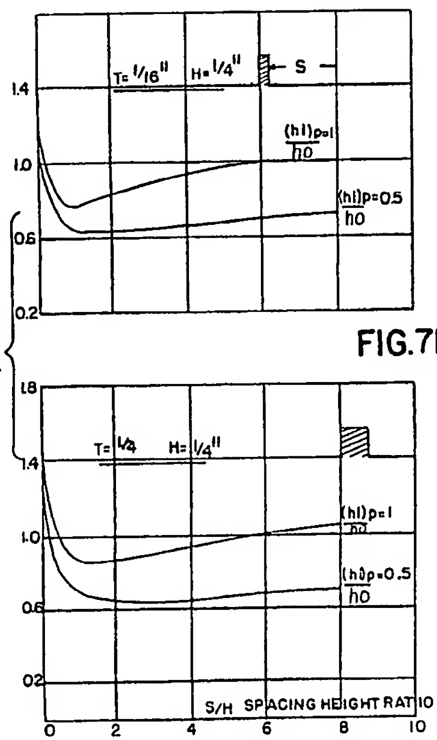


FIG.7A.

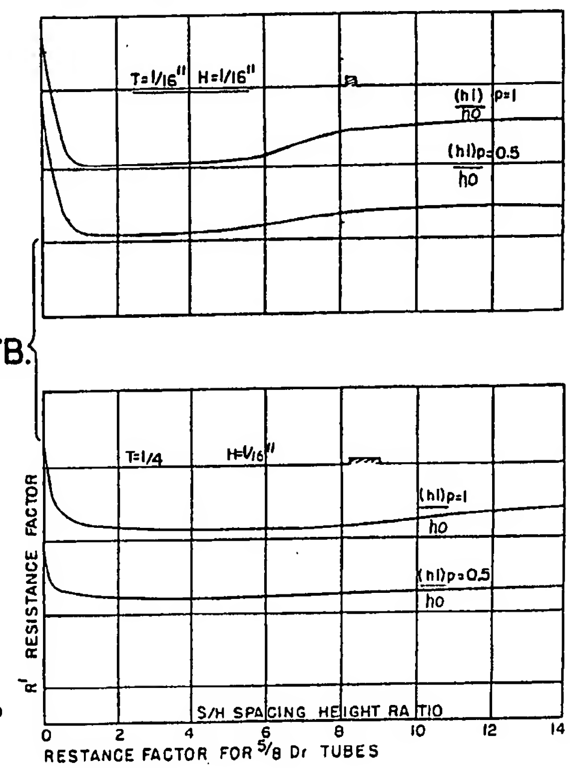
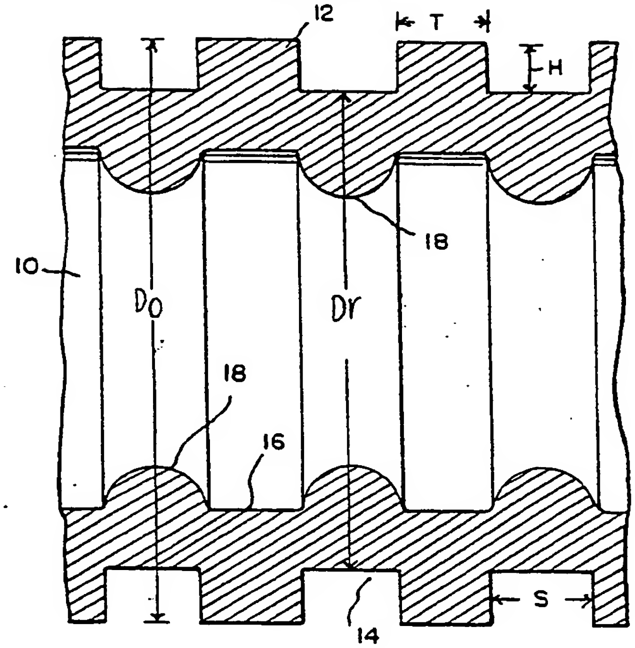
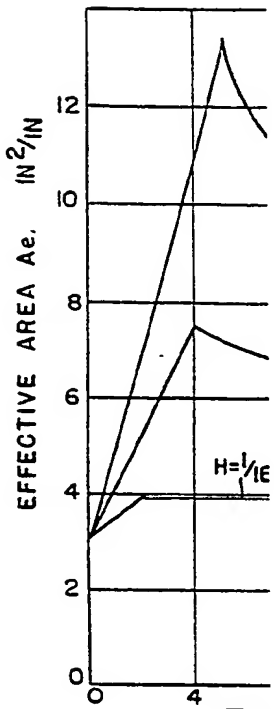


FIG.7B.

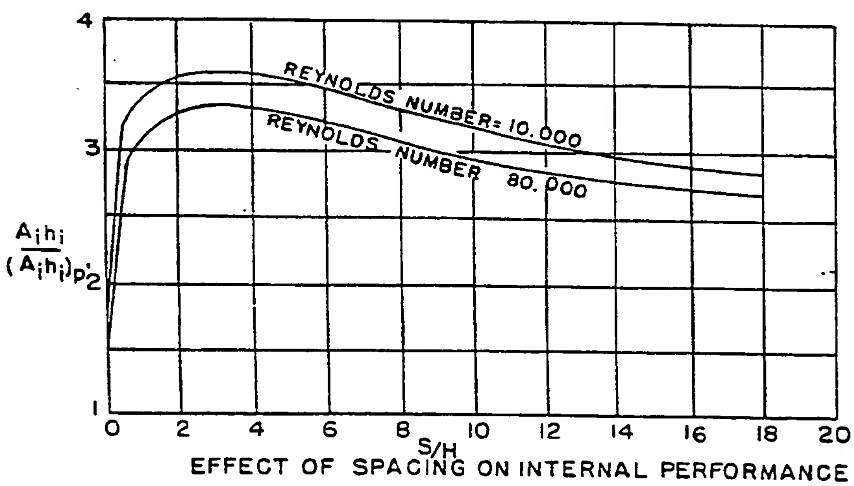
FIG.4.





FI

FIG.6.



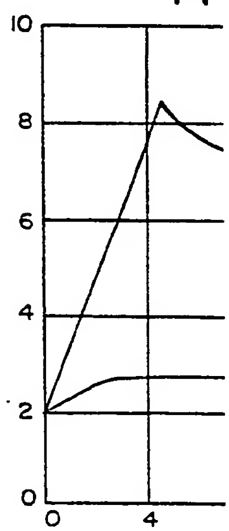


FIG.5

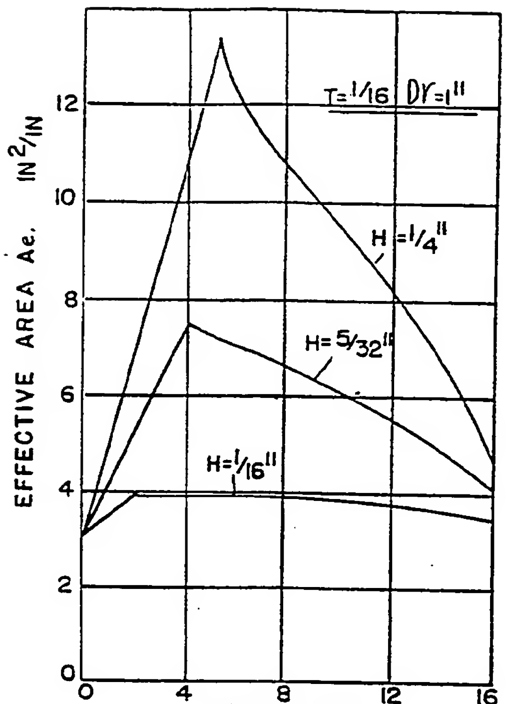


FIG.5A.

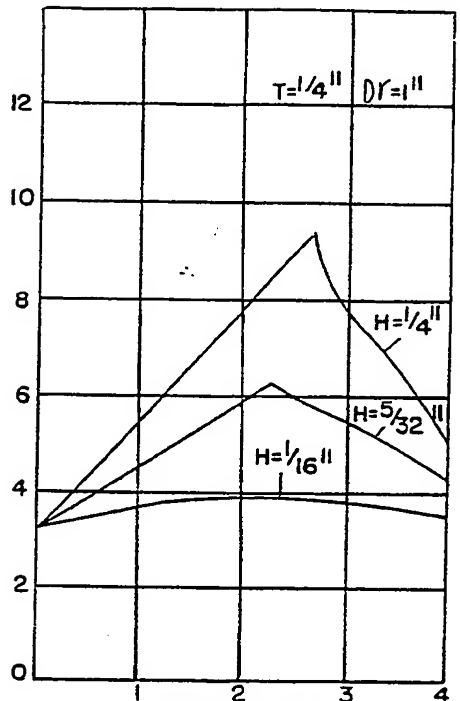


FIG.5B.

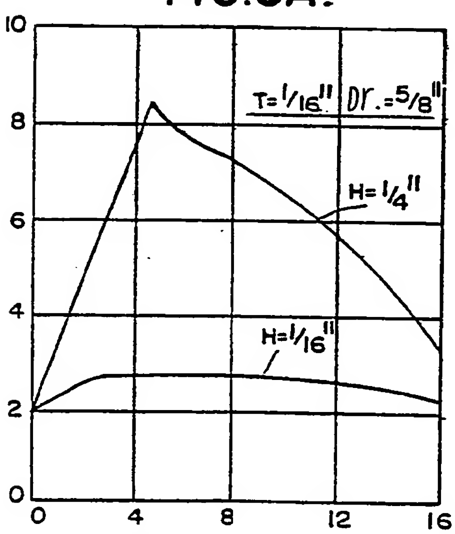


FIG.5C.

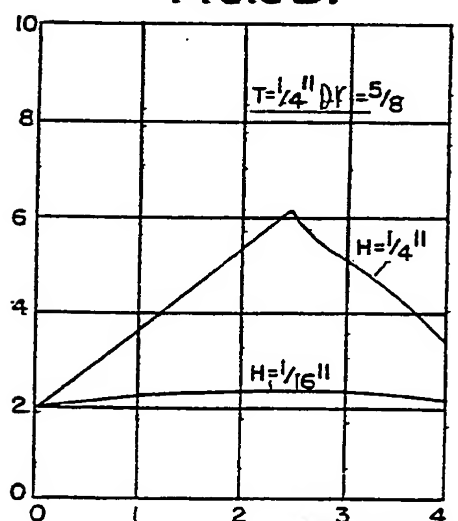


FIG.5D.

FIN FREQUENCY NUMBER OF FINS. PER. INCH

944723 COMPLETE SPECIFICATION

5 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 2

FIG. 4.

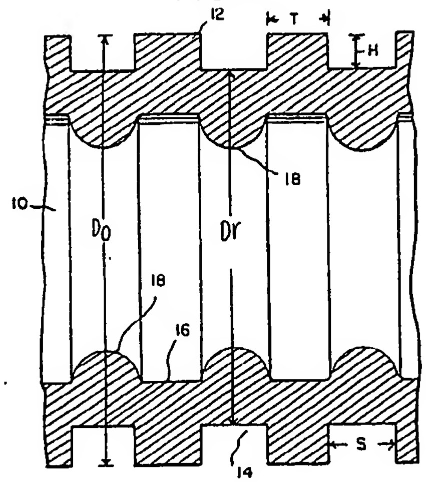
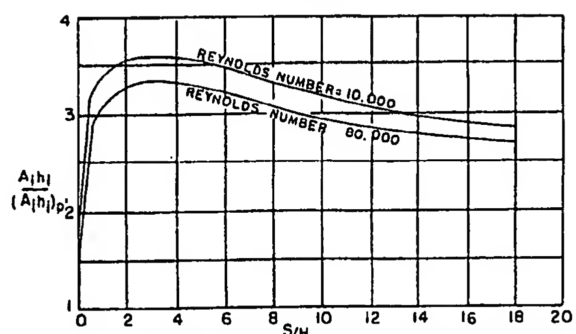


FIG. 6.



EFFECT OF SPACING ON INTERNAL PERFORMANCE

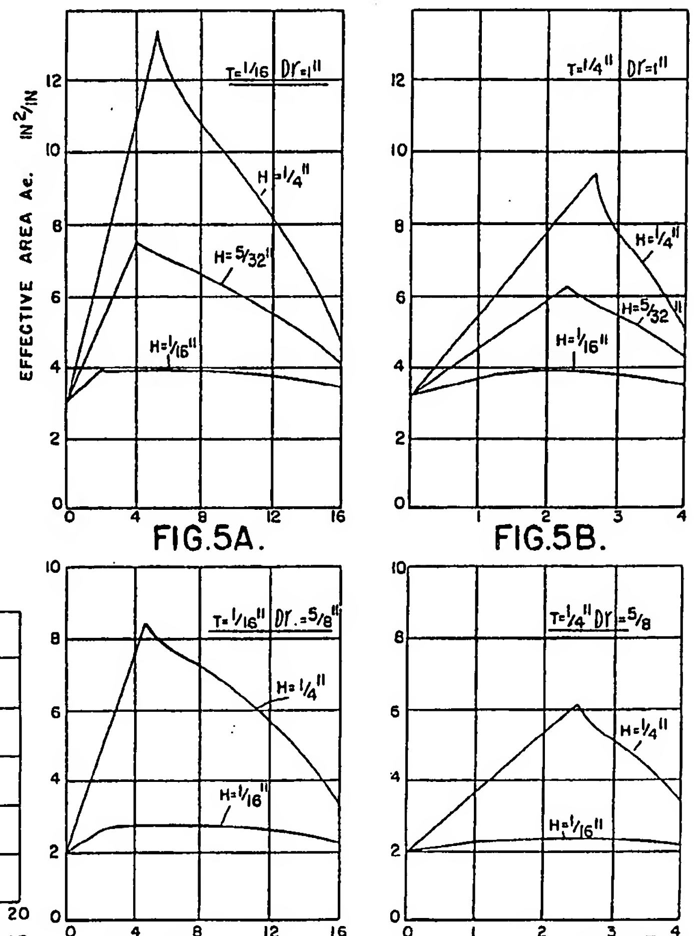


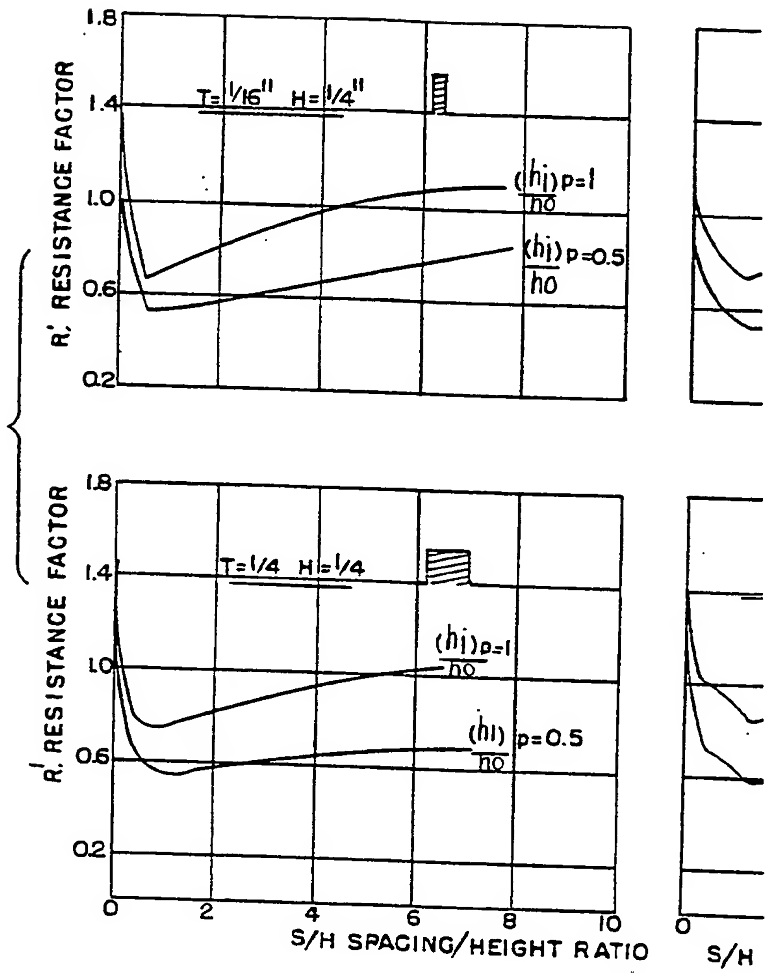
FIG. 5A.

FIG. 5B.

FIG. 5C.

FIG. 5D.

FIG.7C.


R', RESISTANCE FACTOR
T=1/16" H=1/4"
(hi/ho)p=1
(hi/ho)p=0.5
T=1/4 H=1/4
1.8
1.4
1.0
0.6
0.2
0
2
4
6
8
10
S/H SPACING/HEIGHT RATIO
S/H

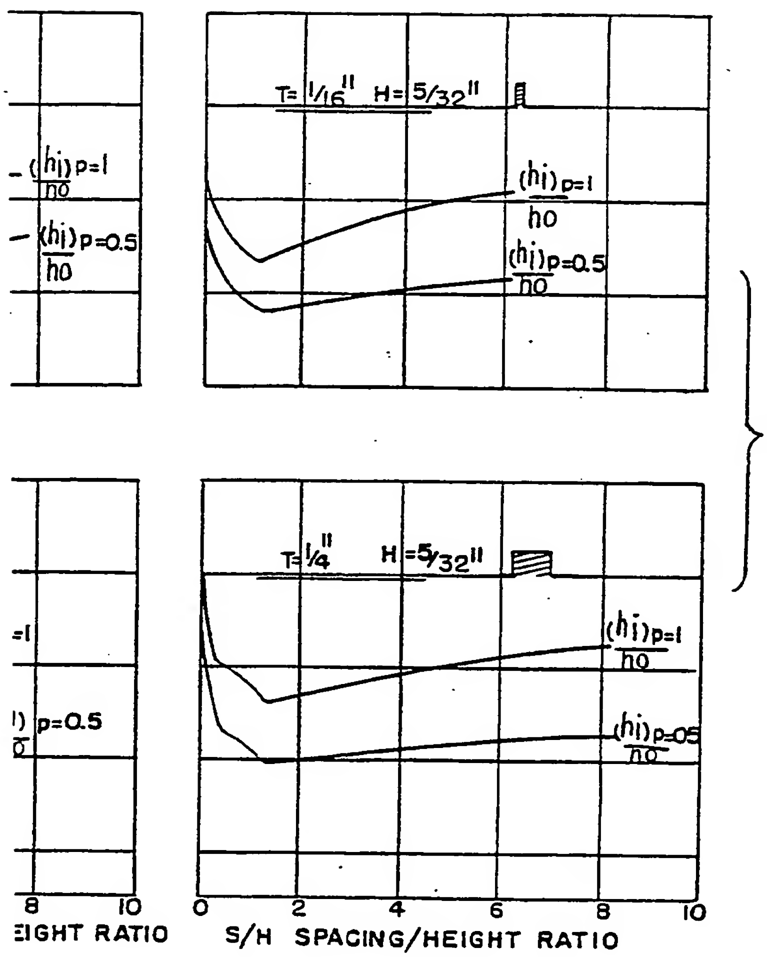

(hi/ho)p=1
(hi/ho)p=0.5
T=1/16" H=5/32"
T=1/4" H=5/32"
(hi/ho)p=0.5
8
10
EIGHT RATIO
0
2
4
6
8
10
S/H SPACING/HEIGHT RATIO

FIG.7D.

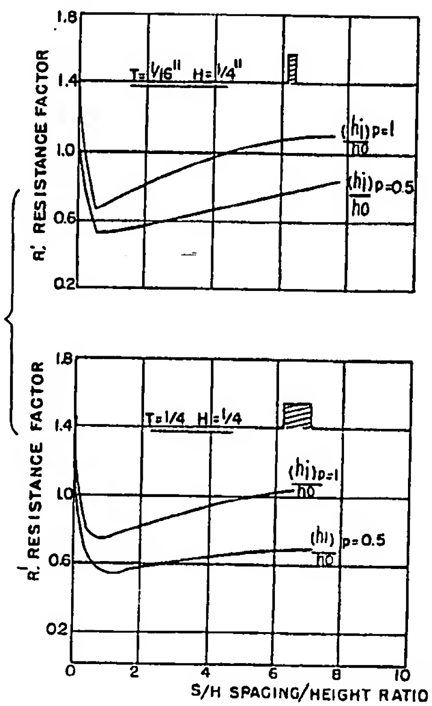

T=1/16" H=1/4"
(hi)p=1 / ho
(hi)p=0.5 / ho
T=1/4 H=1/4
(hi)p=1 / ho
(hi) p=0.5 / ho
R' RESISTANCE FACTOR
1.8
1.4
1.0
0.6
0.2
0
2
4
6
8
10
S/H SPACING/HEIGHT RATIO

FIG.7C.

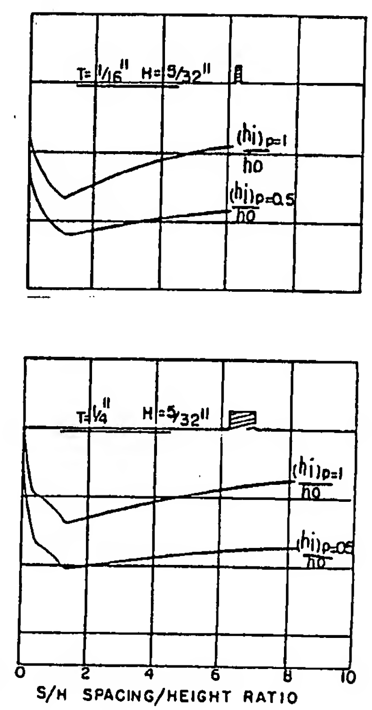

T=1/16" H=5/32"
(hi)p=1 / ho
(hi)p=0.5 / ho
T=1/4" H=5/32"
(hi)p=1 / ho
(hi)p=0.5 / ho
0
2
4
6
8
10
S/H SPACING/HEIGHT RATIO

FIG.7D.

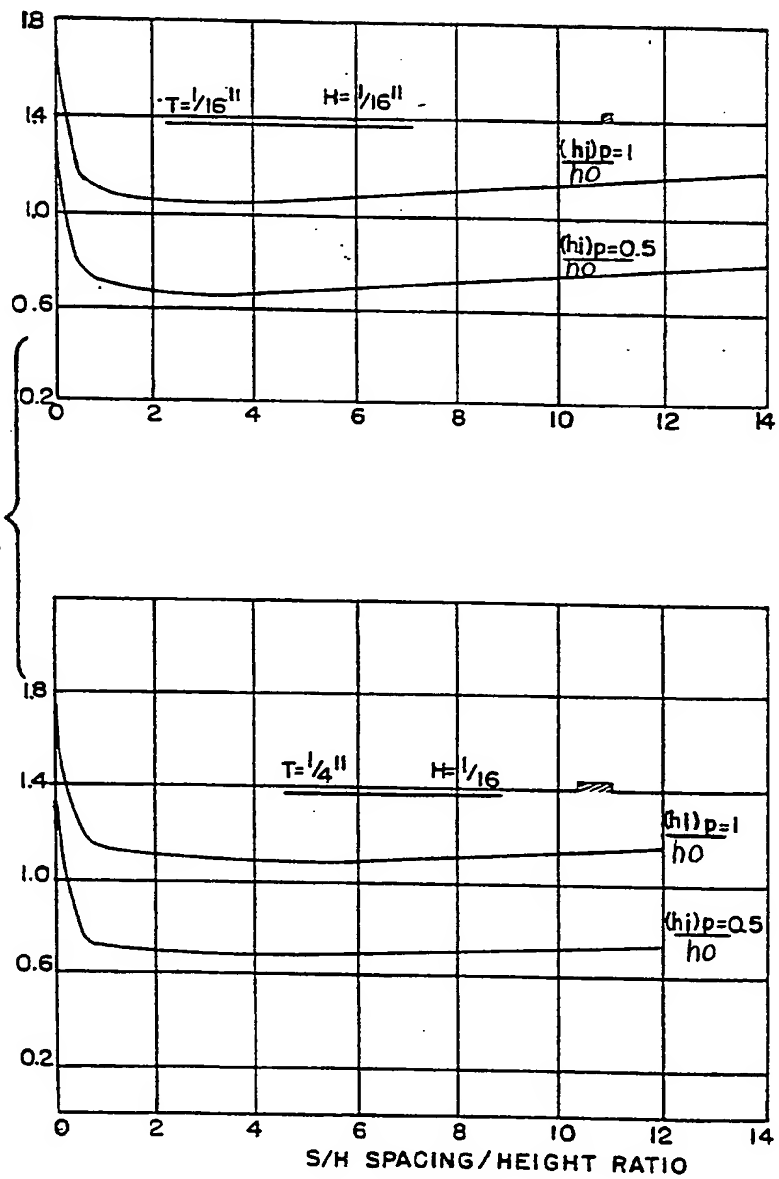

1.8
1.4
1.0
0.6
0.2
0
2
4
6
8
10
12
14
T=1/16"
H=1/16"
(hi)p=1 / ho
(hi)p=0.5 / ho
T=1/4"
H=1/16
S/H SPACING/HEIGHT RATIO

FIG.7E.

FI

944723 COMPLETE SPECIFICATION

5 SHEETS *This drawing is a reproduction of the Original on a reduced scale*

Sheet 5

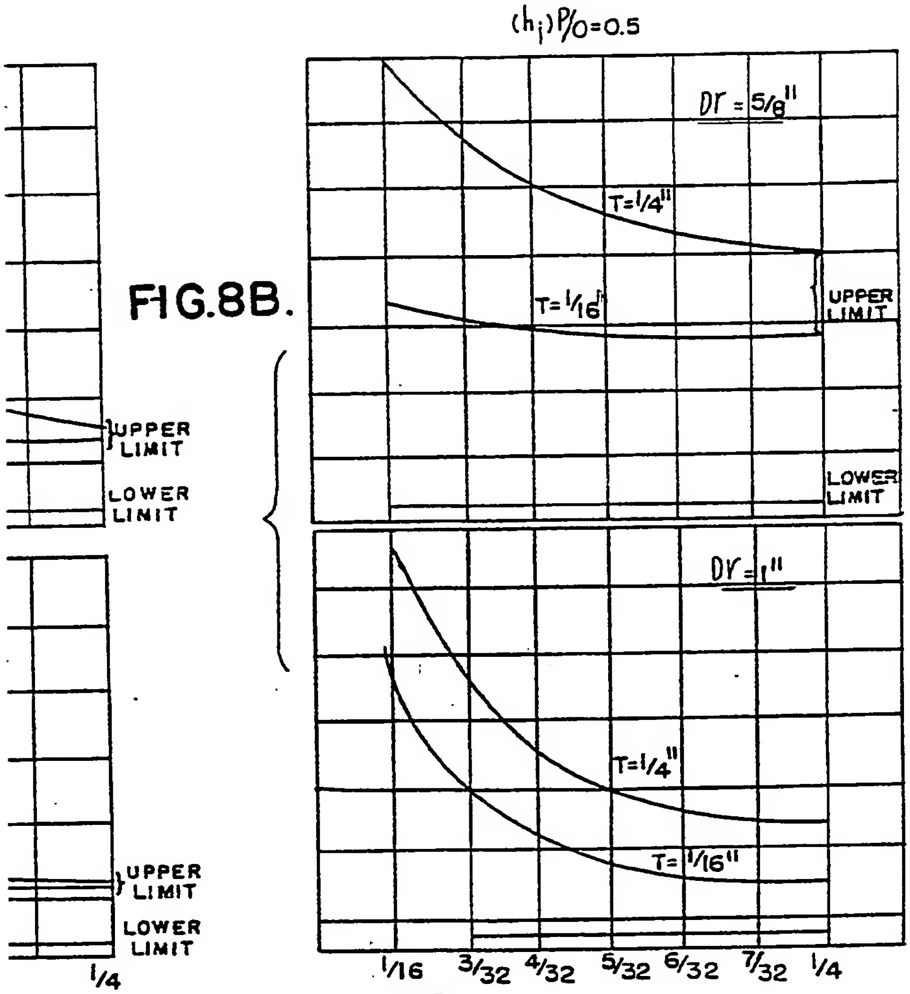


FIG.8B.

OPTIMUM REGION OF S/H

FIG.8A.

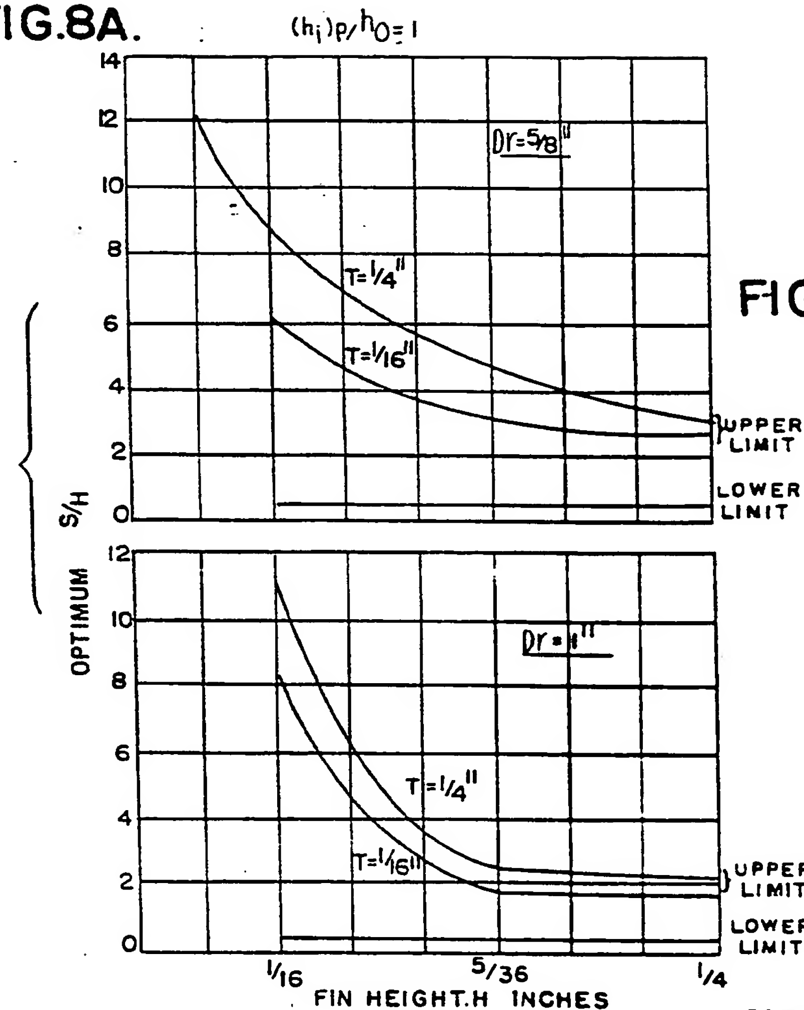


FIG.8E

OPTIMUM

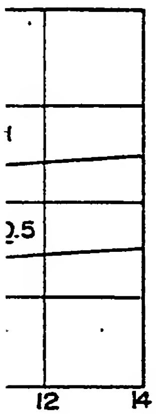
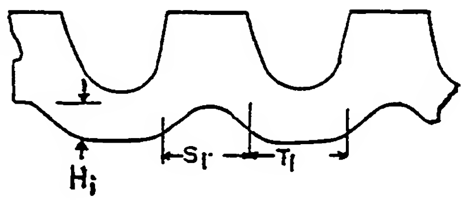


FIG. 9.

FIG. 10.

FIG. 11.



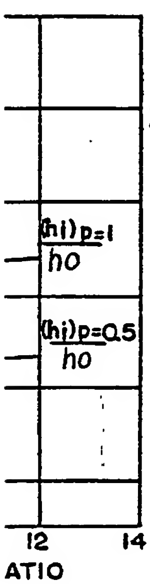


FIG. 12.

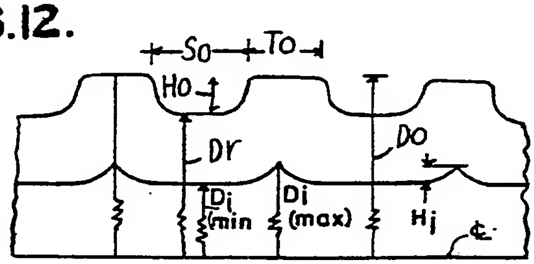


FIG. 13.

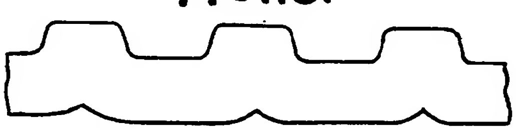
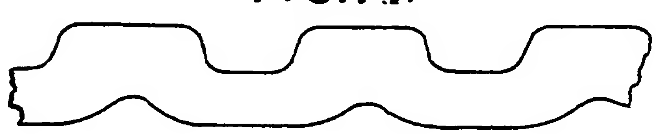


FIG. 14.



944723 COMPLETE SPECIFICATION
5 SHEETS
This drawing is a reproduction of the Original on a reduced scale
Sheet 4

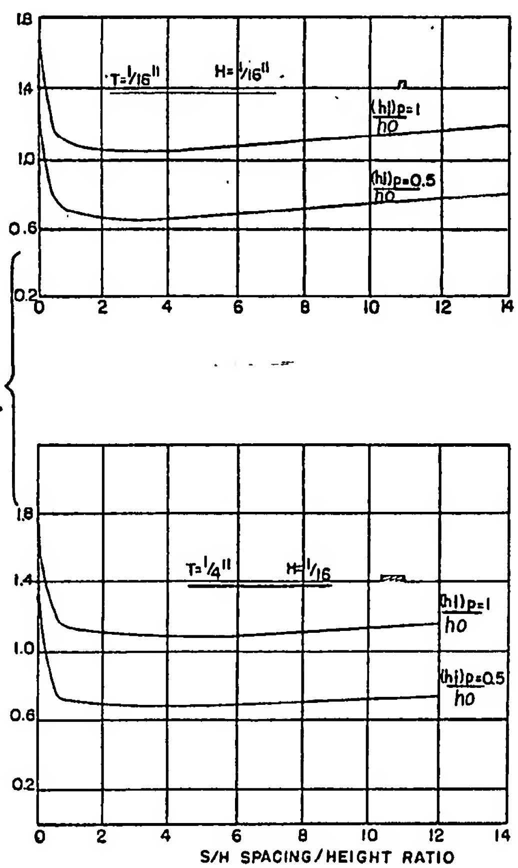


FIG.7E.

FIG.9.


FIG.10.

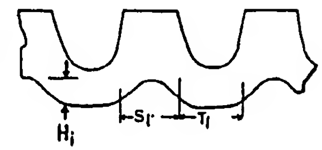

FIG.11.


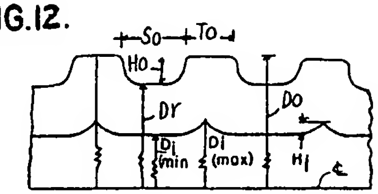
FIG.12.


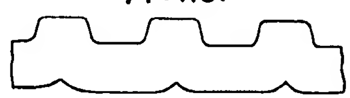
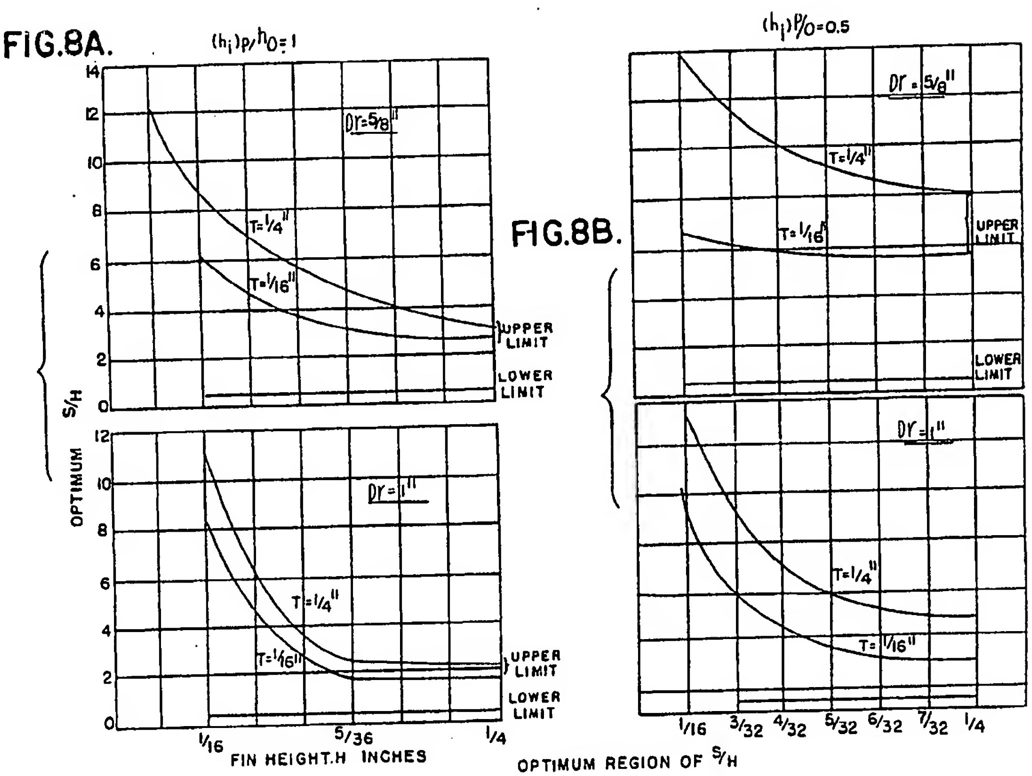
FIG.13.


FIG.14.


FIG.8A. FIG.8B.


$(h_i)_P/h_O = 1$
$(h_i)_P/_O = 0.5$
Dr = 5/8"
Dr = 1"
T = 1/4"
T = 1/16"
UPPER LIMIT
LOWER LIMIT
OPTIMUM S/H
14
12
10
8
6
4
2
0
1/16
5/36
1/4
FIN HEIGHT.H INCHES
3/32
4/32
5/32
6/32
7/32
OPTIMUM REGION OF S/H